RM SE8C12



[redacted]



RM No. SE8C12



4-3-48

SE8C12

NACA

# RESEARCH MEMORANDUM

for the

Air Materiel Command, U. S. Air Force

## PERFORMANCE OF COMPRESSOR OF XJ-41-V TURBOJET ENGINE

## VI - ANALYSIS OF COMPRESSOR FLOW CHOKING

By John W. R. Creagh and Ambrose Ginsburg

Flight Propulsion Research Laboratory
Cleveland, Ohio



TECHNICAL EDITING WAIVED

NATIONAL ADVISORY COMMITTEE
FOR AERONAUTICS

WASHINGTON

MARCH 23 1948

[redacted]

NATIONAL ADVISORY COMMITTEE FOR AERONAUTICS

RESEARCH MEMORANDUM

for the

Air Materiel Command, U.S. Air Forces

# PERFORMANCE OF COMPRESSOR OF XJ-41-V TURBOJET ENGINE

# VI - ANALYSIS OF COMPRESSOR FLOW CHOKING

By John W. R. Creagh and Ambrose Ginsburg

## SUMMARY

An extended analysis was made of the previously reported performance investigation of the original compressor from the XJ-41-V turbojet engine and a similar compressor revised to obtain a 33-percent increase in the geometric passage area at the vaned-collector entrance. This analysis was based on the concept of the vaned-collector entrance as the throat section of a nozzle.

Because of nonuniform air distribution at the vaned-collector entrance, approximately 90 percent of the available flow area was utilized in the original compressor and 94 percent in the revised compressor. The increase in maximum weight flow obtained with the revised compressor was disproportionate to the increased effective critical throat area because the air density at the revised vaned-collector entrance for maximum flow was lower than that obtained in the original compressor. This reduction in density resulted from the large pressure losses near the impeller inlet of the revised compressor, which is indicative of impending flow choking in the impeller. The calculated maximum corrected weight-flow capacity of a compressor consisting of the revised vaneless diffuser and vaned collector with a theoretical impeller that combined peak impeller pressure ratio and peak impeller efficiency at the maximum flow point would be 112 pounds per second for an equivalent impeller speed of 11,500 rpm.

## INTRODUCTION

At the request of the Air Materiel Command, U.S. Air Forces, an investigation of the XJ-41-V turbojet-engine compressor was conducted at the NACA Cleveland laboratory. The objectives were to determine the performance of the compressor over a range of speeds and weight flows and to obtain fundamental information on the aerodynamic problems of large centrifugal compressors. It was shown in references 1 and 2 that the compressor air-flow choking point occurred in the vaned collector for both the original and the revised compressor. It was also shown in reference 2 that, because of the increase in maximum air flow resulting from the increased vaned-collector entrance area of the revised compressor, pressure losses indicative of flow choking occurred in the impeller near the inlet.

The present report is concerned with an elaboration of the previous analysis of the cause of flow choking in the vaned collector and introduces the concept of the vaned-collector entrance as the throat section of a nozzle. The continuity and the steady-flow energy equations as applied to a nozzle are used in an attempt to correlate the compressor maximum-flow performance over a range of impeller speeds. The related effect of impeller performance characteristics on compressor maximum flow is brought out in a discussion of the calculated performance that might be obtained from a theoretical compressor consisting of the revised vaneless diffuser and vaned collector with an impeller capable of producing at maximum flow the peak pressure ratio and peak efficiency of the impeller in the revised compressor. The performance data for the original and the revised compressor presented in references 1 and 2 were used in the analysis presented herein.

## ANALYSIS

The exact location of the choking point in the vaned collector is a function of the flow conditions and the flow-passage geometry. The choking point occurred at or near the entrance to the vaned collector (references 1 and 2). If it is assumed that the flow from the vaneless diffuser enters the vaned collector tangent to the passage walls and that the flow process takes place isentropically, the flow restriction would occur at the minimum passage area.

By assuming the vaned-collector entrance to be a nozzle and by using values of pressures and temperatures corresponding to maximum air flow at several corrected impeller speeds for

both the original and the revised compressor, it would be possible to determine the effective critical throat or nozzle area and to determine whether the geometric throat area could be used as an index of the flow limitations of the compressor. The steady-flow isentropic energy equation for a nozzle can be expressed

$$W = 145.8 \frac{A_2 P_1}{\sqrt{T_1}} \sqrt{\left(\frac{p_2}{P_1}\right)^{2/\gamma} - \left(\frac{p_2}{P_1}\right)^{\frac{\gamma+1}{\gamma}}} \qquad (1)$$

where

$W$ air weight flow, pounds per second
$P_1$ total pressure at nozzle entrance, inches mercury absolute
$p_2$ static pressure at nozzle throat, inches mercury absolute
$T_1$ total temperature at nozzle entrance, $^\circ$F absolute
$A_2$ nozzle throat area, square feet
$\gamma$ ratio of specific heats

The maximum weight flow through the nozzle will be reached when the critical pressure occurs in the throat section and equation (1) reduces to

$$W_c = 37.6 \frac{A_2 P_1}{\sqrt{T_1}} \qquad (2)$$

where $W_c$ is the maximum weight flow. Because the air-flow data presented in references 1 and 2 are corrected to standard sea-level pressure and temperature at the compressor inlet, the same procedure will be followed herein. Equation (2) is then expressed

$$(W_c)_s = \frac{49.5\ (P_1/P_0)\ A_2}{\sqrt{T_1/T_0}} \qquad (3)$$

where

$(W_c)_s$ maximum weight flow corrected to standard sea-level temperature and pressure, pounds per second

$T_0$ compressor inlet total temperature, $^{\circ}F$ absolute
$P_0$ compressor inlet total pressure, inches mercury absolute

and $A_2$ is hereinafter designated as the effective critical throat area.

In applying equation (3) to the data of references 1 and 2, $T_1$ is the total temperature measured in the simulated burner annulus and $P_1$ is the total pressure at the impeller tip calculated according to the method given in reference 1. The results of several check runs at maximum-flow conditions with a total-pressure probe in the vaneless diffuser near the vaned-collector entrance of both the original and the revised compressor showed that a negligible difference existed between the calculated impeller-discharge pressures and the pressure-probe readings. Table I shows the experimental values of temperatures, pressures, and maximum air flows used to calculate the effective critical throat areas of the original and the revised compressors at several equivalent impeller speeds. The compressor pressure ratio is included in the table as a convenient reference to the data in references 1 and 2.

Figure 1 shows the approximate passage-area variation through a section of the vaned collector formed by planes perpendicular to a geometric mean flow path. The area values shown are the totals for the 12 vaned-collector passages. The minimum geometric area occurred at the vane entrance; a value of 0.51 square foot was obtained for the original compressor and 0.68 square foot for the revised compressor.

## RESULTS AND DISCUSSION

The calculated effective critical throat areas for the original and revised compressors at maximum weight flow over a range of equivalent impeller speeds from 7,000 to 11,500 rpm are shown in figure 2. A constant effective critical throat area of 0.46 square foot was obtained for the original compressor and 0.64 square foot for the revised compressor regardless of variations in impeller speed and maximum weight flow. These values approximate the values of geometric throat areas shown in figure 1 and indicate that the conception of the vaned-collector entrance as the throat of a nozzle is valuable in establishing a useful index of the ultimate air-flow capacity of this compressor. For each compressor a constant percentage of the geometric throat area was available to the flow when sonic velocity was reached at the vaned-collector

entrance. As a result of nonuniform air-flow distribution at the vaned-collector entrance, the original compressor utilized approximately 90 percent of the geometric area and the revised compressor, approximately 94 percent.

The ratio of the maximum corrected weight flows of the revised compressor to the maximum corrected weight flows of the original compressor is shown in figure 3 over a range of equivalent impeller speeds from 7,000 to 10,000 rpm together with the ratio of the effective critical throat areas of the two compressors. The weight-flow-ratio curve is considerably lower than the area-ratio curve at 7000 rpm and the difference between the two curves increases rapidly as the speed is increased. This fact together with a study of equation (3) suggested that either the total-pressure ratios or the total-temperature ratios at the throat of the revised vaned collector had been adversely affected at some point in the compressor system prior to the throat section.

Figure 4 shows the impeller-discharge total-pressure ratios and total-temperature ratios for both compressors over a range of equivalent impeller speeds at maximum flow. The curves show slightly lower impeller-discharge temperature ratios for the revised compressor but these are accompanied by large reductions in impeller-discharge total-pressure ratios. The difference between the impeller-discharge total-pressure ratios of the two compressors was comparatively small at the low speeds but increased rapidly as the speed increased. At an equivalent impeller speed of 10,000 rpm, the impeller-discharge total-pressure ratio for the revised compressor was 78 percent of that obtained for the original compressor. These reductions in total pressure were due to pressure losses in the impeller near the impeller inlet at high flows and explain why the increase in weight flow with the revised compressor was disproportionate to the increased effective critical throat area (fig. 3).

Figure 5 shows the maximum corrected weight flow obtained over a range of equivalent impeller speeds for the original compressor, the revised compressor, and a theoretical compressor consisting of the revised vaned collector and vaneless diffuser with a hypothetical impeller. This impeller component was assumed to have, at maximum flow, the peak imepller pressure ratio and peak impeller efficiency obtained experimentally on the revised compressor. From equation (3), the pressures and temperatures corresponding to peak impeller efficiency and pressure ratio, and the effective critical throat area of 0.64 square foot (fig. 2), the maximum corrected weight flows were determined for the theoretical compressor. At the

rated speed of 11,500 rpm with the theoretical compressor, the weight-flow capacity could be increased from 76 pounds per second with the revised compressor to 112 pounds per second, a 47-percent increase. This optimum value of 112 pounds per second at 11,500 rpm would require that the impeller have the extraordinarily high specific flow capacity value of 12,300 cubic feet per minute per square foot. Any attempt to increase the pressure ratio and maximum weight flow of the XJ-41-V compressor must include modifications to the impeller component to eliminate the pressure losses therein.

## SUMMARY OF RESULTS

From an analysis of the compressor flow-choking characteristics based on the concept of the vaned-collector entrance as the throat section of a nozzle, the following results were obtained:

1. A constant effective critical throat area of 0.46 square foot was obtained for the original compressor and 0.64 square foot for the revised compressor regardless of impeller speed and weight flow. These values approximate the geometric throat areas and indicate that the nozzle concept can be used as an index of the ultimate air-flow capacity of this compressor.

2. Owing to nonuniform flow distribution at the vaned-collector entrance, the original compressor utilized approximately 90 percent of the available geometric passage throat area and the revised compressor utilized approximately 94 percent.

3. Above an equivalent impeller speed of 7000 rpm, the ratio of the maximum weight flows of the revised and orignal compressors were less than the ratio of effective critical throat areas of the two compressors and the difference increased with an increase in impeller speed.

4. The increase in maximum weight flow of the revised compressor was disproportionate to the increased effective critical throat area because of large pressure losses in the impeller near the impeller inlet. In any attempt to increase the pressure ratio and maximum weight flow of the compressor, the impeller must be modified to eliminate these losses.

5. A theoretical compressor consisting of the revised vaned collector and vaneless diffuser with an impeller advantageously combining peak impeller pressure ratio and peak

impeller efficiency at maximum flow would be capable of producing a maximum corrected air flow of 112 pounds per second at rated speed of 11,500 rpm.

Flight Propulsion Research Laboratory,
National Advisory Committee for Aeronautics,
Cleveland, Ohio, March 12, 1948.

John W R Creagh

John W. R. Creagh,
Aeronautical Research
Scientist.

Ambrose Ginsburg

Ambrose Ginsburg,
Aeronautical Research
Scientist.

Approved:

Robert O. Bullock,
Aeronautical Research
Scientist.

Oscar W. Schey,
Aeronautical Research
Scientist.

mjz

REFERENCES

1. Creagh, John W. R., and Ginsburg, Ambrose: Performance of Compressor of XJ-41-V Turbojet Engine. IV - Performance Analysis over Range of Compressor Speeds from 5000 to 10,000 rpm. NACA RM No. SE7L12, Army Air Forces, 1948.

2. Ginsburg, Ambrose, Creagh, John W. R., and Michel, Donald: Performance of Compressor of XJ-41-V Turbojet Engine. V - Performance Analysis of Compressor with Revised Vaned Collector over Range of Compressor Speeds from 3600 to 11,500 rpm. NACA RM No. SE8A22, Army Air Forces, 1948.

TABLE I - MAXIMUM-FLOW DATA USED IN DETERMINING EFFECTIVE

CRITICAL THROAT AREAS OF VANED COLLECTOR

| Compressor | Equivalent impeller speed (rpm) | Compressor total-pressure ratio | Compressor inlet total pressure, $P_0$ (in. Hg abs.) | Compressor inlet total-temperature, $T_0$ (°F abs.) | Impeller total pressure ratio, $P_1/P_0$ | Impeller total temperature ratio, $T_1/T_0$ | Corrected maximum weight flow, $(W_c)_s$ (lb/sec) |
|---|---|---|---|---|---|---|---|
| Original | 7,000 | 1.41 | 13.9 | 528 | 1.98 | 1.24 | 40.8 |
| | 8,000 | 1.81 | 14.0 | 534 | 2.38 | 1.31 | 47.4 |
| | 9,000 | 1.52 | 14.0 | 541 | 2.88 | 1.38 | 55.7 |
| | 10,000 | 2.11 | 13.9 | 541 | 3.38 | 1.48 | 63.3 |
| Revised | 7,000 | 1.36 | 14.4 | 541 | 1.86 | 1.21 | 52.0 |
| | 8,000 | 1.59 | 14.1 | 535 | 2.13 | 1.28 | 59.9 |
| | 9,000 | 1.77 | 13.9 | 533 | 2.43 | 1.35 | 66.5 |
| | 10,000 | 2.00 | 14.0 | 541 | 2.65 | 1.43 | 70.9 |
| | 11,500 | 2.29 | 5.8 | 428 | 3.01 | 1.58 | 76.1 |

945

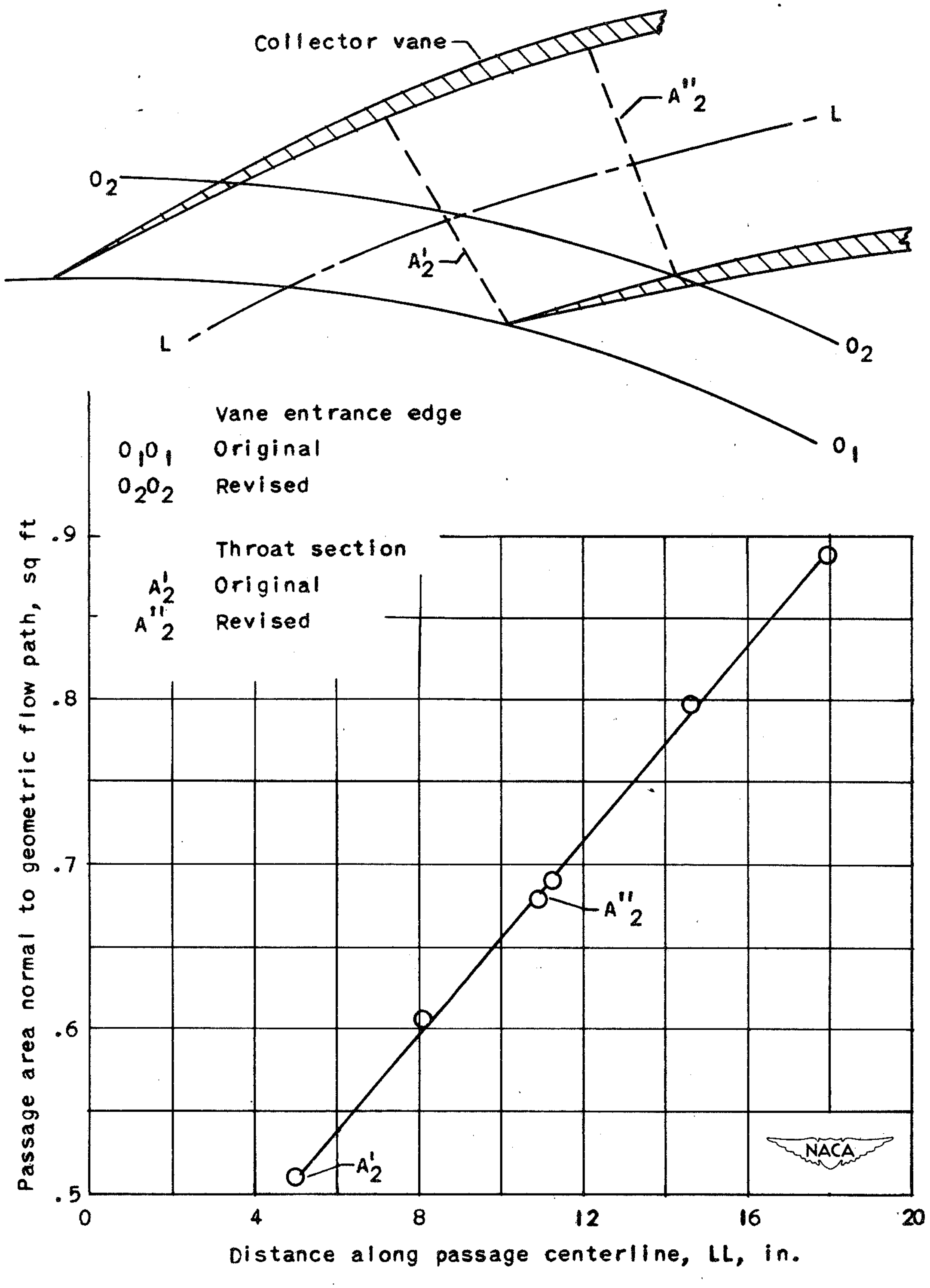


Figure 1. - Variation of passage area normal to geometric mean flow path with distance along vaned collector centerline.

945

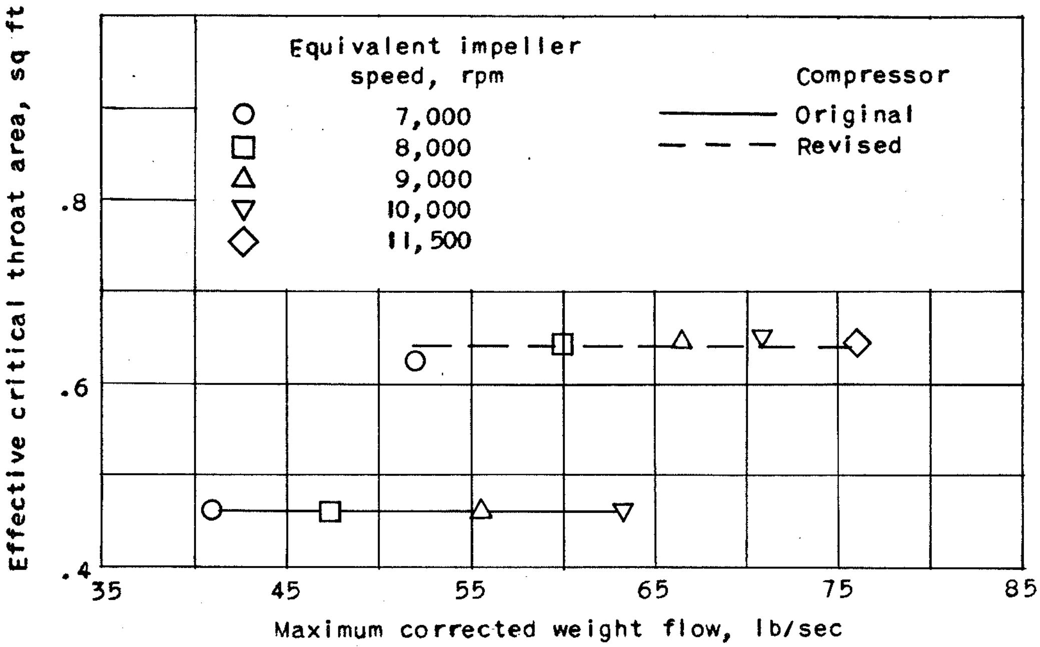


Figure 2. - Variation of effective critical throat area with maximum corrected weight flow for original and revised compressor.

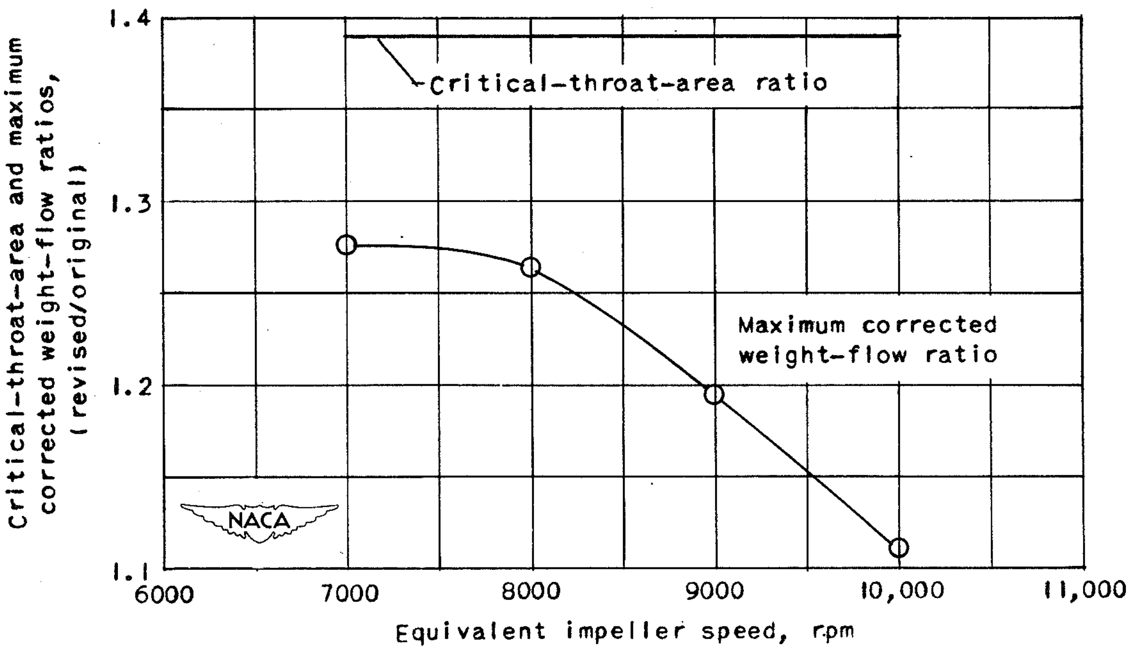


Figure 3. - Comparison of increase in critical throat area with increase in maximum corrected weight flow for original and revised compressor.

945

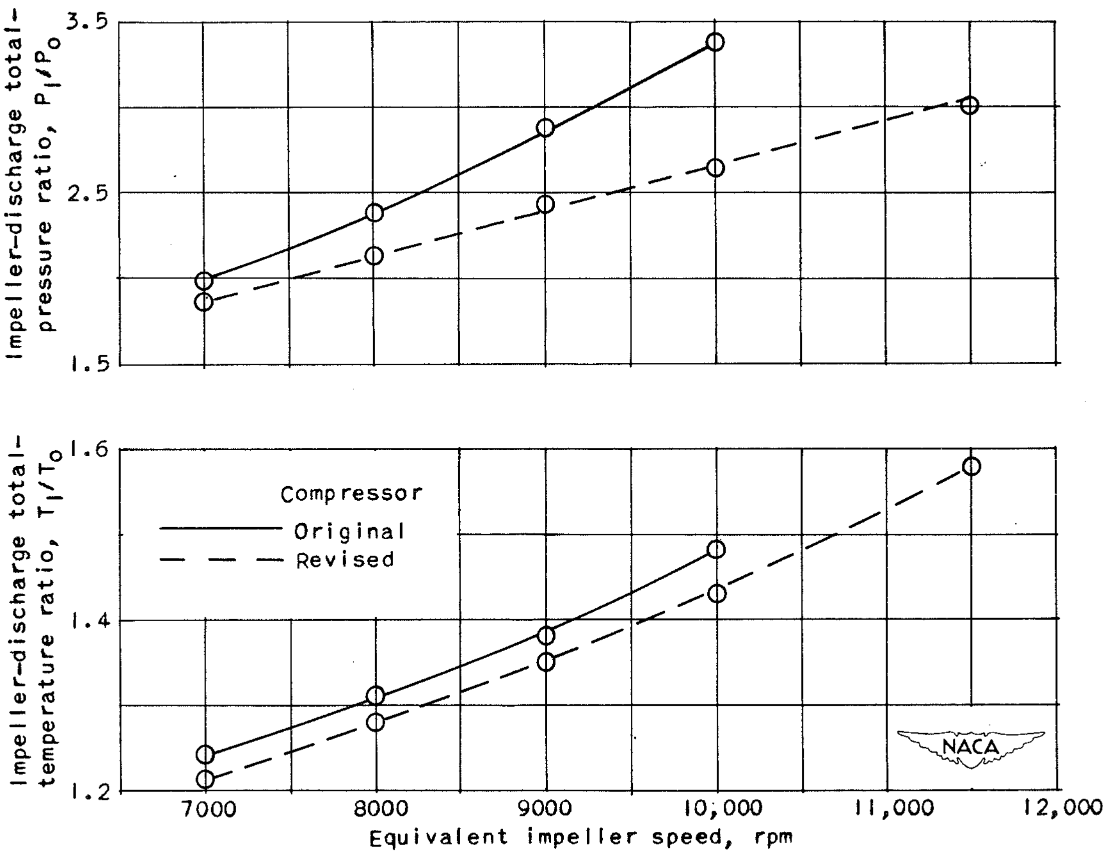


Figure 4. - Comparison of impeller-discharge total-temperature ratio and total-pressure ratio for original and revised compressor.

945

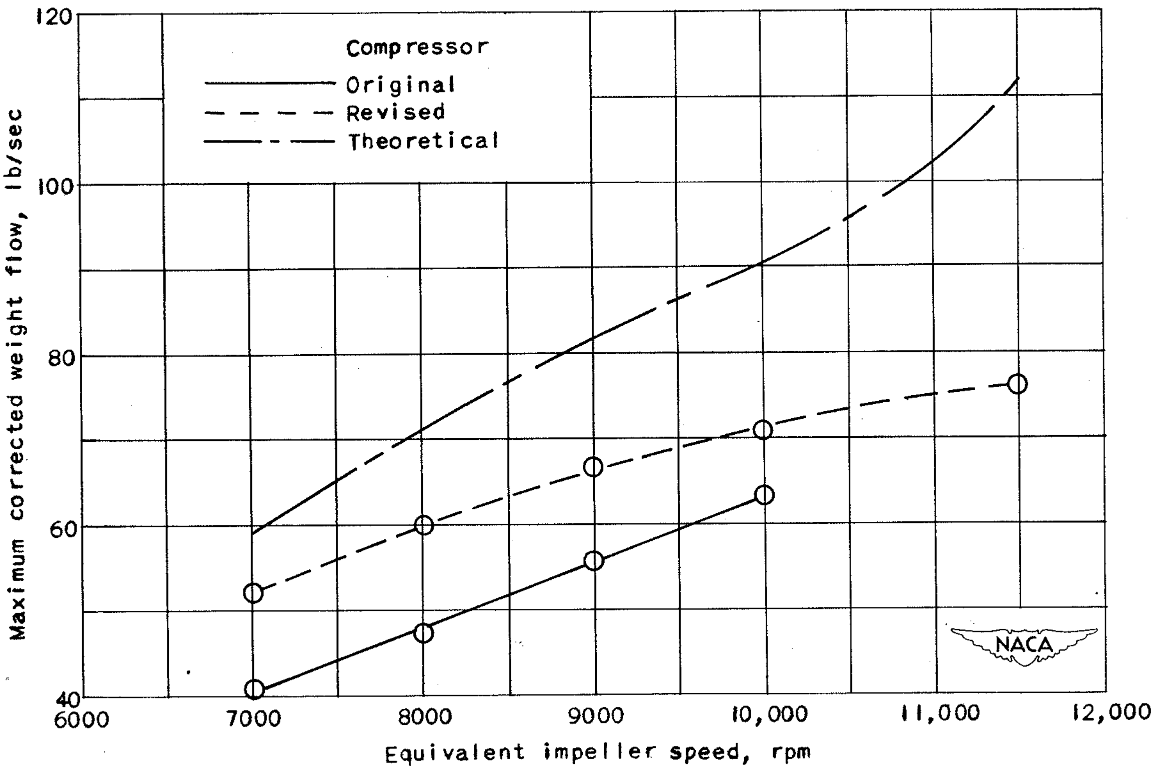


Figure 5. - Comparative values of maximum corrected weight flow for three compressors.

UNCLASSIFIED

UNCLASSIFIED